operating with a Margin of Stability of zero *at the Threshold of Stability*. This is demonstrated by the fact that both the Direct and Quadrature Dynamic Stiffnesses are zero at the whirl or whip frequency.

This can also be illustrated by a Root Locus plot [3]. Figure 10 shows a Root Locus plot of the rotor system under test. The operating points on the plot were obtained by first obtaining rotor parameters from the Dynamic Stiffness curve fits and applying them to a rotor model that included the Fluid Inertia Effect [4]. For the whip data, the Root Locus point could not be determined from a model, so it was plotted based on observed behavior. The Root Locus plot displays the experimental rotor system natural frequency on the vertical axis versus a growth or decay factor on the horizontal axis. The entire right half-plane represents an operating region where rotor system is forbidden, that is, rotor vibration, once it started, would increase in amplitude forever. For normal, stable operation, the rotor system must operate in the left half-plane.

In Figure 10, note that the operating points for stable operation are in the left half-plane. The operating points for whirl and whip are located *on the vertical axis*, and the natural frequency of the rotor system *is the same as the whirl and whip frequency*. Thus, in whirl and whip the rotor system is at the Threshold of Stability and it is precessing *at the rotor system natural frequency*.

Summary

These experiments have demonstrated that the Margin of Stability steadily decreases as the rotor speed approaches the Threshold of Stability. As the rotor enters unstable operation, it starts whirling or whipping at its natural frequency. In both whirl and whip, the Dynamic Stiffness zero crossings occur at the frequency of the whirl or whip.

We can conclude that a rotor system operating in a fluid instability will have zero Direct and zero Quadrature Dynamic Stiffness at the frequency of the instability.

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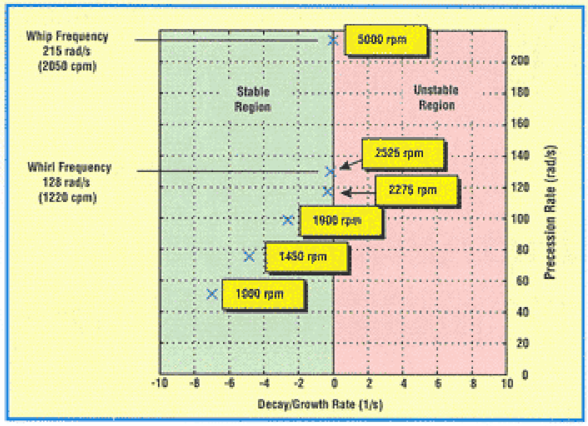

Figure 10. Root Locus plot of the operating points in the experiments. This graph plots Decay/Growth rate on the horizontal axis versus rotor system natural frequency on the vertical axis. Stable rotor operation plots in the left half-plane. Rotor operation in the right half-plane is forbidden, since it would imply unlimited rotor vibration amplitude. The rotor operation, stable at speeds of 1900 rpm and below, plots very close to the vertical axis at 2275 rpm. The rotor in whirl at 2525 rpm is precessing at the Low Eccentricity Natural Frequency, while in whip at 5000 rpm it is precessing at the High Eccentricity Natural Frequency.

The difference between whirl and whip

Obviously, there is a difference between the behavior of the rotor system in whirl versus its behavior in whip. To understand why this is so, it is first necessary to realize that the rotor system direct stiffness *K*, actually consists of a combination of the shaft spring and the bearing spring. (There can be additional stiffness contributions from other sources, but for the purposes of this article we can neglect them.) The shaft spring stiffness is effectively in series with the bearing spring. When two springs are in series, then the weakest spring controls the overall stiffness of the combination.

At the onset of whirl, the rotor was operating in the center of the bearing where the bearing spring stiffness (which increases strongly near the bearing wall) was minimum. The rotor system stiffness was controlled by the weaker bearing stiffness. Thus, the natural frequency of the rotor system was at a minimum, which is called the *Low Eccentricity Natural Frequency* (Figure 2). As the rotor starts into whirl, the diameter of the orbit and the bearing stiffness increased. This stiffness increase changed the zero point of the Direct Dynamic Stiffness slightly. Thus, the stiffness increase keeps the rotor system on the edge of the stability threshold and prevents the orbit diameter from increasing. The natural frequency of the rotor also increases slightly. This holds true as long as the rotor speed remains the same. If, however, the rotor speed increases, the rotor exceeds the new Threshold of Stability, the orbit diameter increases, and the cycle repeats. Thus, in whirl, it is the changing bearing spring stiffness that keeps increasing the natural frequency of the rotor system, and the whirl frequency tracks along an order line on the spectrum cascade plot as rotor speed increases. This order line is approximately λX.

However, at some point, the bearing spring stiffness becomes so high that it is no longer the weakest spring in the rotor system. The shaft spring is now the weakest spring, and the shaft spring cannot be modified. Thus, the rotor system asymptotically approaches what we call the *High Eccentricity Natural Frequency* (Figure 2). In this region, the subsynchronous frequency of precession of the rotor system remains constant, and we define this instability region as whip. It is this frequency that often correlates with the "name-plate critical" in rotating machinery.